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EFFECT OF THE THERMODYNAMIC PROPERTIES OF THE WORKING MEDIUM ON THE CHOICE OF OPTIMUM PARAMETERS FOR A GAS TURBINE

by O. N. Yemin and N. N. Bykov

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Translation of "Vliyaniye termodinamicheskikh svoystv rabochego tela
na vybor optimal'nykh parametrov gazovoy turbiny."
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ABSTRACT

Comparative criteria are discussed for deciding the optimum parameters (load factor, coefficient of discharge, flow deflection angles) of conventional hydrocarbon fuel gas turbines, as well as turbines operating on helium, hydrogen, and other working media, as a function of the thermodynamic properties of the latter. Larger values are recommended for the load factor and coefficient of discharge when the characteristic velocity of sound of the medium is high, as in the case of H and He. The problem of rectifying the exit flow from the final stage is considered; this is best achieved by limiting the load factor of the last stage or installing rectifying vanes at the exit. In either case, a penalty is paid in overall efficiency, but the second expedient is better suited to working media with a high characteristic sonic velocity.

1. INTRODUCTION AND STATEMENT OF THE PROBLEM /65*

The problem of choosing the optimum parameters for a gas turbine whose working medium consists of the combustion products of hydrocarbon fuels in air has been rather thoroughly developed in the technical literature. The basic analytical parameters, including the load per stage, reactive effectiveness, etc., are chosen so as to provide acceptable Mach numbers M in the flow-through section, hence high efficiency with a reasonable number of stages (ref. 1).

The numerical values of the parameters depend on the velocity of sound in the gas and the work done in expansion for a given pressure drop and temperature drop, i.e., on the properties of the working medium.

It is sensible, in this connection, to utilize the cumulative experience already gained in the design of turbines that operate with the combustion products of hydrocarbon fuels in air. In the ensuing discussion, with reference to the type of working medium used in these turbines, the latter will be called simply "gas turbines," as opposed to "helium," "hydrogen," or other designations.

We will adopt as our fundamental thermodynamic characteristic of the working medium, governing the velocity level in the flow-through section of the turbine, the coefficient in the equation for the critical velocity of sound:

$$\frac{a_{cr}}{\sqrt{T_0^*}} = \sqrt{\frac{2k}{k+1} gR}.$$

*Numbers in the margin indicate pagination in the original foreign text.

For the gases investigated in the present paper (comprising those most suited for use as working media), this quantity is equal to 16.15 for argon, 18.1 for "gas" (from conventional fuel), 22.5 for neon, 51.2 for helium, and 69.6 for hydrogen (ref. 3).

We propose to explain the physical aspect of how to approach the choice of optimum parameters, by comparing the conventional gas stage and helium stage of a turbine. The work done in expansion for helium at the same initial temperature and the degree of variation in pressure is several times greater than for gas, and the design of helium turbines with loads typical of gas stages leads to a substantial increase in the number of stages.

Without considering in detail the fundamental postulates of the theory of /66
similarity of the processes occurring in bladed machines, we will directly compare the velocity triangles for the elementary stages of various turbines (the notation and terminology are discussed in section 2).

The velocity triangles for a typical, fairly heavily loaded elementary stage of gas turbine ($k = 1.33$) with $\rho_t = 0.35$, $\bar{H}_{ti} = 2.0$, $\bar{C}_a = 1.0$ are shown in figure 1a.

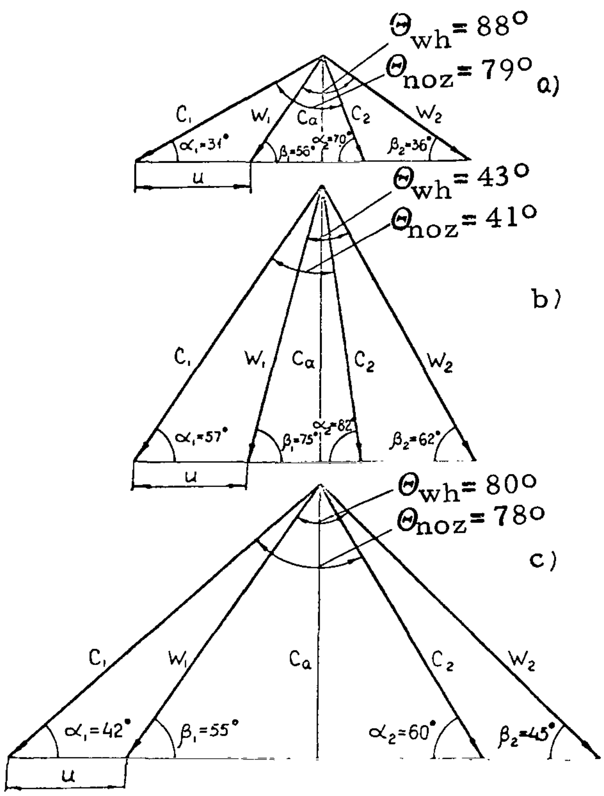

Figure 1. Velocity Vector Triangles for the Elementary Stages of Various Turbines

For U = 273 m/sec and $T_0^* = 1200°K$, these values of the velocity triangle parameters correspond to $\lambda_{1c} = 0.94$; $\lambda_{1w} = 0.59$; $\lambda_{2c} = 0.505$.

If we build a helium turbine stage with the same velocity diagram and the same value of T_0^*, the λ numbers in the flow-through section will be approximately one third less. From the viewpoint of similarity theory, it is best in the design of a helium stage to increase all of the vectors in the velocity diagram proportionately by a factor of three, so that the level of the M numbers (or λ numbers) in the helium stage will be the same as in the gas stage. This approach, however, is unrealistic, as the required tripling of the peripheral velocity is unsuitable, in fact sometimes quite impossible for strength considerations.

However, a helium stage can be designed with the same value of U, $\overline{H}_{ti}$, and ρ_t as its gas counterpart (fig. 1a), but with large $\overline{C}_a = 2.5$. As shown in fig- /67
ure 1b, this kind of stage will be characterized by much smaller values of the flow deflection angles in the rotor and nozzle assembly. If it is realized with $T_0^* = 1200°K$, fairly moderate values will be obtained for the coefficients λ in the flow-through section ($\lambda_{1c} = 0.506$; $\lambda_{2c} = 0.483$), since the velocity of sound in helium is considerably higher than in gas. Consequently, the stage can be driven up to loads at which the flow deflection angles are still acceptable and ensure high efficiencies. The stage of a helium turbine with $\overline{C}_a = 2.5$, $\rho_t = 0.35$, U = 273 m/sec, $\overline{H}_{ti} = 4.16$ and, consequently, with a theoretical work more than twice as great as for the stages indicated above, is shown in figure 1c. In this case, $T_0^* = 1200°K$, $\lambda_{1c} = 0.635$, $\lambda_{2c} = 0.500$, i.e., the indicated parameters lie with permissible limits for gas turbines.

This comparison shows that a turbine operating on a working medium with a high velocity of sound permits higher values to be chosen for $\overline{C}_a$ and $\overline{H}_{ti}$ by comparison with conventional gas turbines.

In the present paper, we investigate the problem of choosing optimum $\overline{C}_a$ and $\overline{H}_{ti}$ as a function of the properties of the working medium. Our fundamental criterion for rating the suitability of the choice of $\overline{C}_a$ and $\overline{H}_{ti}$ will be the efficiency of the turbine stage, calculated on the basis of the results of blow-through on flat blading. The effects of the peripheral velocity, which exerts a considerable influence on the size, weight, and strength of the turbine stage, has not been investigated.

However, in selecting the fundamental parameters of the stage it is necessary to make a comprehensive evaluation of their influence on the blade height, as well as the size, weight, strength, technological, and other characteristics of the turbine stage.

2. EFFECT OF $\overline{C}_a$ and $\overline{H}_{ti}$ ON THE EFFICIENCY OF A TURBINE STAGE

We propose to examine a certain intermediate stage in a multistage turbine, so that the velocity at the inlet to the nozzle assembly and at the exit from the stage has the same magnitude and direction, i.e., $C_0 = C_2$. Furthermore, we will suppose for simplicity that $C_{1a} = C_{2a} = C_a$. Under these conditions, it is sufficient to specify three independent dimensionless parameters of the stage in order to uniquely define the configuration of the velocity triangle, i.e., all the angles, including the flow deflection angles in the elements of the stage, Θ_{noz} and Θ_{wh} (fig. 1).

We select the following parameters:

coefficient of discharge: $$\overline{C}_a = \frac{C_a}{U},$$

load factor: $$\overline{H}_{ti} = \frac{\Delta C_u}{U} = \frac{C_{1u} + C_{2u}}{U},$$

Kinematic reactive effectiveness: $$\rho_T = 1 - \frac{C_{1u} - C_{2u}}{2U}.$$

We use $\overline{H}_{ti}$ and ρ_t to determine the relative (dimensionless) values of the peripheral velocity components of the flow in the flow-through section: /68

$$\overline{C}_{1u} = \frac{C_{1u}}{U} = \frac{\overline{H}_{ti}}{2} + (1 - \rho_T),$$
$$\overline{C}_{2u} = \frac{C_{2u}}{U} = \frac{\overline{H}_{ti}}{2} - (1 - \rho_T).$$

If, in addition, the value of the coefficient of discharge $\overline{C}_a$ is given, the directions of the flow in the flow-through section of the stage are defined, for example:

$$\operatorname{tg} \alpha_1 = \frac{\overline{C}_a}{\overline{C}_{1u}} = \frac{\overline{C}_a}{\frac{\overline{H}_{ti}}{2} + (1 - \rho_T)},$$

$$\operatorname{tg} \beta_1 = \frac{\overline{C}_a}{\overline{C}_{1u} - 1} = \frac{\overline{C}_a}{\frac{\overline{H}_{ti}}{2} - \rho_T}.$$

Consequently, the flow deflection angles in the nozzle assembly and wheel will be uniquely determined by the chosen values of $\overline{H}_{ti}$, ρ_t, and $\overline{C}_a$.

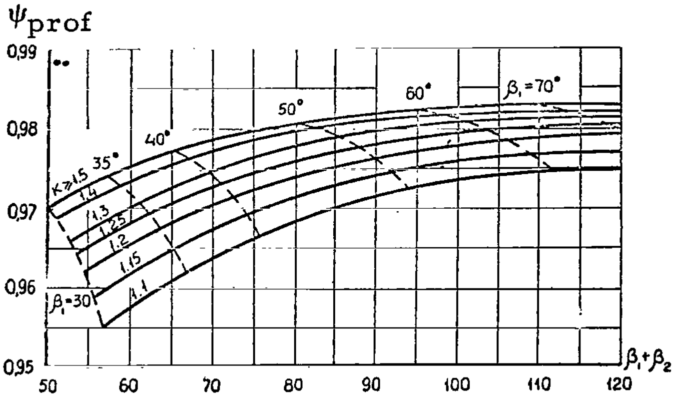

Figure 2. Profile Loses in Flat Turbine Blading.

As is known, the profile losses in turbine blading at subsonic speeds in the flow-through part are determined primarily by the type of blading and the flow deflection angle in it, but almost not at all by the velocity. The data shown in figure 2 give the dependence of the velocity coefficient (ψ in the rotor blading, ϕ in the nozzle blading) on the type of blading as defined by the parameter $K = \sin\beta_1/\sin\beta_2$ and on the flow deflection angle in it, $\Theta = 180 - (\beta_1 + \beta_2)$ (for the nozzle blading, the corresponding angles α_2 and α_1 are used). These graphs were constructed on the basis of the experimental data of G. Yu. Stepanov and V. L. Epshteyn on the mean frictional losses in blading with a calculated flow inlet angle (the tip losses in this case were determined /69
from the formula $\zeta_{cr} = 0.01/\sin\beta_2$).

If we assume that the results obtained in the blow-through of air on the blading can be extended to the flow of other working media past the blading (also at subsonic velocities),[1] then the loss factors and efficiencies of the turbine will depend only on $\overline{H}_{ti}$, ρ_t, and $\overline{C}_a$, as long as the velocities in the flow-through section of the turbine are subsonic.

For the relative blade heights in current usage, it is customary to choose an analytic mean diameter $\rho_t = 0.25$ to 0.35. Therefore, in order to simplify the analysis, all subsequent calculations were carried out for a fixed value of the reactive effectiveness, $\rho_t = 0.3$. In this case, the efficiency is determined only by the load factors $\overline{H}_{ti}$ of the simple turbine stage and the axial velocity $\overline{C}_a$. The results of a calculation of the efficiency in terms of delayed parameters as a function of $\overline{H}_{ti}$ and $\overline{C}_a$, taking into account only profile losses, are shown in figure 3.

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[1]This assumption, in correspondence with the requirements of similarity theory, is in need of experimental verification.

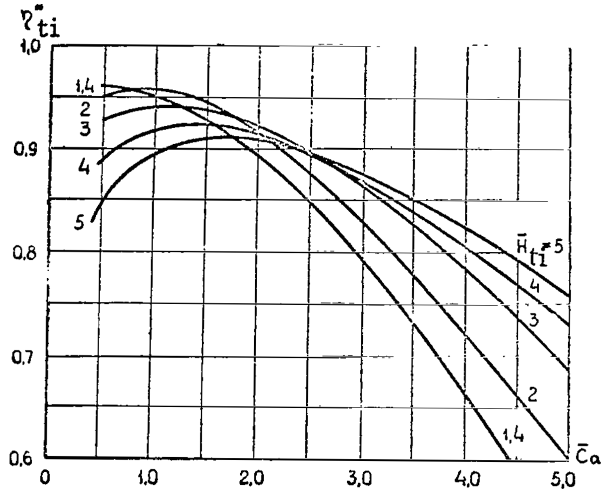

Figure 3. Effect of the Coefficient of Discharge and Load Factor of a Simple Turbine Stage on its Efficiency (Calculations Performed for $\rho_t = 0.3$).

An analysis of this graph shows that the choice of higher coefficients of discharge $\overline{C}_a$ dictates an increase in the optimum value of the load factor for the turbine stage, i.e., the value for which maximum stage efficiency is obtained. However, the efficiency of the turbine (fig. 4) decreases with increasing $\overline{C}_a$, even with the choice of optimum load factor, because the exit velocity increases. The choice of increased values for the coefficient of discharge will be attended by deterioration of the economy criteria and could only be of interest from the viewpoint of possibly raising the load factor and, hence, decreasing the number of turbine stages.

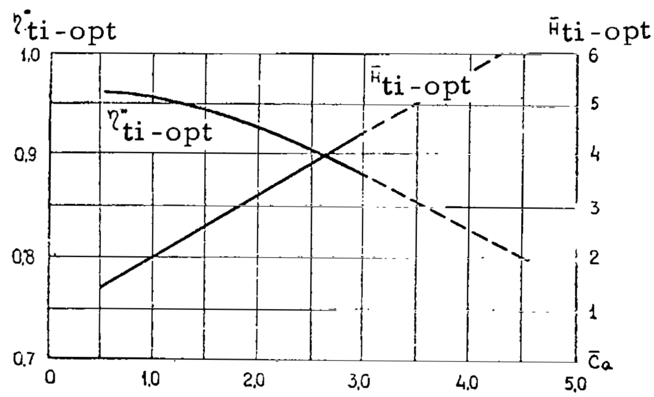

Figure 4. Effect of the Coefficient of Discharge on the Optimum Value of the Load Factor and Efficiency of a Simple Turbine Stage.

Consequently, the graphs in figures 3 and 4 are universal, i.e., they are applicable to all working media, as long as the velocities in the flow-through part of the turbine are subsonic for the chosen values of $\overline{H}_{ti}$ and $\overline{C}_a$. We will now establish the effect of the fundamental parameters of the elementary stage on the velocity levels in the flow-through section of the turbine using different working media. /70

3. EFFECT OF $\overline{H}_{ti}$ and $\overline{C}_a$ ON THE FLOW VELOCITIES

The choice of definite values for $\overline{H}_{ti}$, ρ_t, and $\overline{C}_a$, as already indicated, uniquely determine the configuration of the velocity vector triangle, i.e., the angles in the flow-through section and the relative magnitudes of all its velocities. To find the values of the flow-through velocities, it is necessary to /71 specify the value of the peripheral velocity, or to specify the parameter U/T_0^* in order to determine the λ numbers.

The results of a calculation of the relative velocity coefficient at the inlet to the rotor for $\rho_t = 0.3$, $U = 250$ m/sec, $T_0^* = 900°$, and $\overline{H}_{ti_{opt}}$ and all values of $\overline{C}_a$ are shown in figure 5.

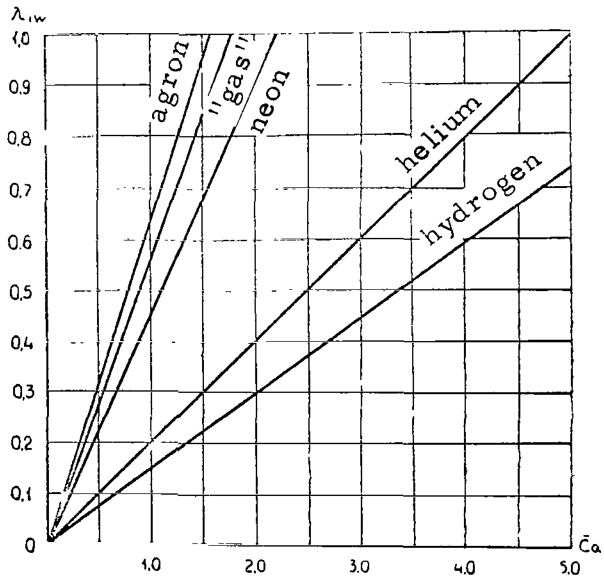

Figure 5. Effect of the Coefficient of Discharge on the Relative Velocity Coefficient at the Inlet to the Rotor of a Turbine with Various Working Media.

It is usually recommended to have $\lambda_{1w} \cong 0.7$. In the design calculations of gas turbines, therefore, $\overline{C}_a \leqslant 1.0$ and, accordingly (fig. 4), and $\overline{H}_{ti} = 1.4$ to 1.7 (and, less frequently, $\overline{H}_{ti} = 2$) is assigned as suitable values for the load factors of the individual stages. Analogous recommendations can also be made for working media such as argon and neon, whose velocities of sound are close to that of "gas."

For helium and hydrogen (fig. 5), the choice of higher values for $\overline{C}_a$ is not accompanied by the onset of sonic velocities in the flow-through section, in addition to which this permits the choice of higher values of $\overline{H}_{ti}$ and, consequently, fewer turbine stages.

The calculations represented in figure 6 for other values of the peripheral velocities and temperatures at the turbine intake with various working media, assuming a value of $\lambda_{1w} = 0.7$, permit the possible value of $\overline{C}_a$ to be estimated.

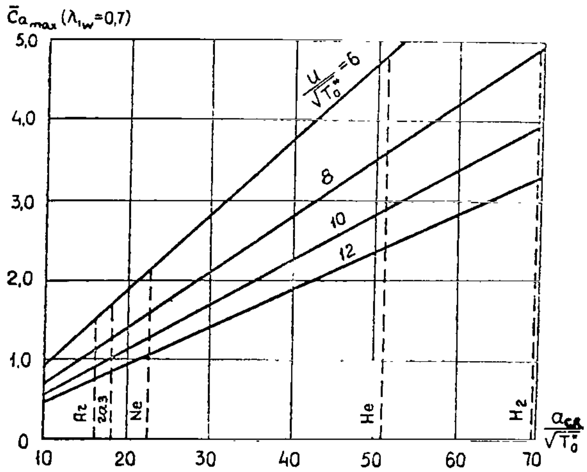

Figure 6. Effect of the Reduced Peripheral Velocity and Properties of the Working Medium on the Maximum Possible Value of the Coefficient of Discharge.

It is evident that for helium and even more so for hydrogen, higher values can be chosen for $\overline{C}_a$ even for large peripheral velocities.

In choosing the fundamental parameters of the turbine stage it is also essential to consider the velocity coefficient λ_{2c} behind the turbine. Its value, of course, determines the efficiency of operation of the turbine aft equipment, and in multistage turbines it also affects the nature of the flow intake to the next nozzle assembly. It is normally assumed in the calculations for gas turbines that $\lambda_{2c} = 0.5$ to 0.6.

Figure 7 gives the results of a calculation of the velocity coefficient behind the stage with U = 250 m/sec, $T_2^* = 875$°K as a function of $\overline{C}_a$, where, as before, $\overline{H}_{ti} = H_{ti_{opt}}$ and $\rho_t = 0.3$. They also indicate the possibility of designing helium and hydrogen turbines with enhanced values of $\overline{C}_a$. /72

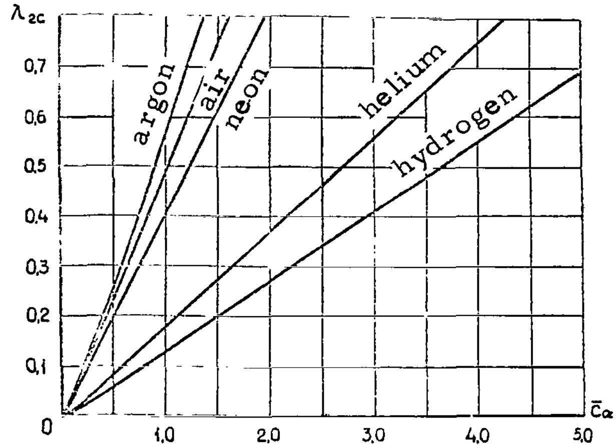

Figure 7. Effect of the Fundamental Turbine Parameters and Properties of the Working Medium on the Value of the Velocity Coefficient Behind the Stage.

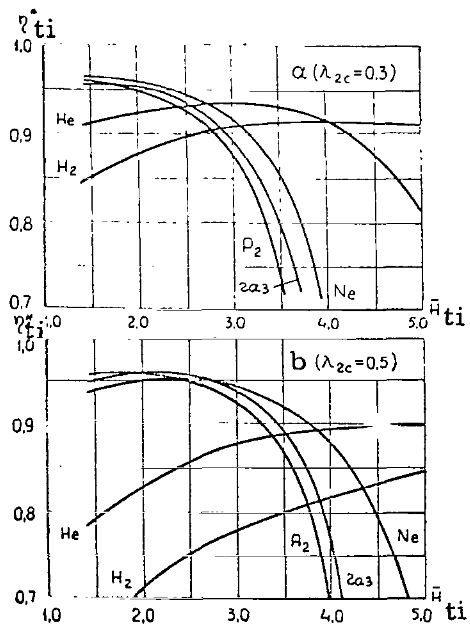

Figure 8. Dependence of Turbine Efficiency for Various Working Media on the Load factor: a) $\lambda_{2c} = 0.3$; b) $\lambda_{2c} = 0.5$.

Sometimes in the practice of performing the design calculations for turbines, their transmissive capacity is not evaluated in terms of the coefficient of discharge, but in terms of quantities related to it, such as λ_{2c} or the flow deflection angle α_1 from the nozzle assembly. The dependence of the stage efficiency of a turbine on the load factors and velocities behind the turbine is shown in figure 8 for various working media; the curves were derived from the universal relation given in figure 3 for the special case U = 250m/sec, $T_2^* = 875°K$. /73

For small $\lambda_{2c} = 0.3$ (fig. 8a), the optimum value of the load factor for a gas turbine is $\overline{H}_{ti} = 1.5$. A helium turbine, even with these values of the turbine exit velocity coefficient, can be designed with $\overline{H}_{ti} = 3.0$, a hydrogen turbine with $\overline{H}_{ti} = 4.0$.

With an increase in λ_{2c}, the optimum values of $\overline{H}_{ti}$ will increase (fig. 8b).

The design of a turbine stage with enhanced $\overline{C}_a$ involves the necessity of building the nozzle assembly with large angles α_1.

It is recommended for gas turbines that the latter angle be chosen as $\alpha_1 = 20$ to $35°$, corresponding to a workable range of $\overline{C}_a = 0.5$ to 1.0. In cases when the working medium is characterized by a high velocity of sound, higher values can be chosen for the angle α_1, corresponding to higher $\overline{C}_a$. The only limitation on the choice of α_1 (or $\overline{C}_a$) is the tendency to avoid too short a blade.

4. APPLICATION OF RECTIFYING VANES BEHIND THE TURBINE

In the design of a turbine stage with a high load factor, it is necessary to bear in mind that the exit flow from the turbine will not be axial. For intermediate stages, this is all right, since the exit flow from the stage will impinge on the intake to the next nozzle assembly. Hence, the recommendation of having $\alpha_2 \geqslant 70°$ at the exit from the intermediate stages of gas turbines (type TRD and TVD turbines) is primarily intended to limit the flow deflection angle in the rotor blading and succeeding nozzle assembly.

In the design of turbines with high $\overline{C}_a$ and $\overline{H}_{ti}$, the angle α_2 can be made somewhat smaller than indicated in the design recommendations for ordinary gas turbines. However, the flow deflection angles in this case (fig. 1c) can have an acceptable value, providing high efficiency.

In selecting the parameters of the last stage of a multistage turbine it is necessary to produce axial (or nearly axial) emergence of the flow from the turbine for effective operation of the system following the turbine. This is accomplished by assigning a moderate value to the load factor of the last stage;

in the case of a turbine stage with enhanced values of this coefficient, rectifying vanes are placed after the exit.

For the TRD and TVD gas turbines, such directing vanes have not been employed, because at the values of $\overline{C}_a$ that can be realized in these turbines an increase in $\overline{H}_{ti}$ produces a decrease in the efficiency inherent in the turbine due to the increased flow deflection angle, not to mention the additional losses in the directing vanes (ref. 2).

In turbines with high values of $\overline{C}_a$, where the optimum value of the load factor corresponds to appreciable spin in the flow behind the rotor, the application of rectifying vanes may prove useful. /74

In order to illustrate this notion, we estimated the efficiency of a turbine with rectifying vanes for the generation of axial exit flow from the stage.

Curves, calculated from data on the losses in diffuser vanes and showing the efficiency of a turbine with rectifying vanes as a function of load factor, are given in figure 9, which also shows for comparison curves calculated previously for the efficiency of a turbine without rectifying vanes.

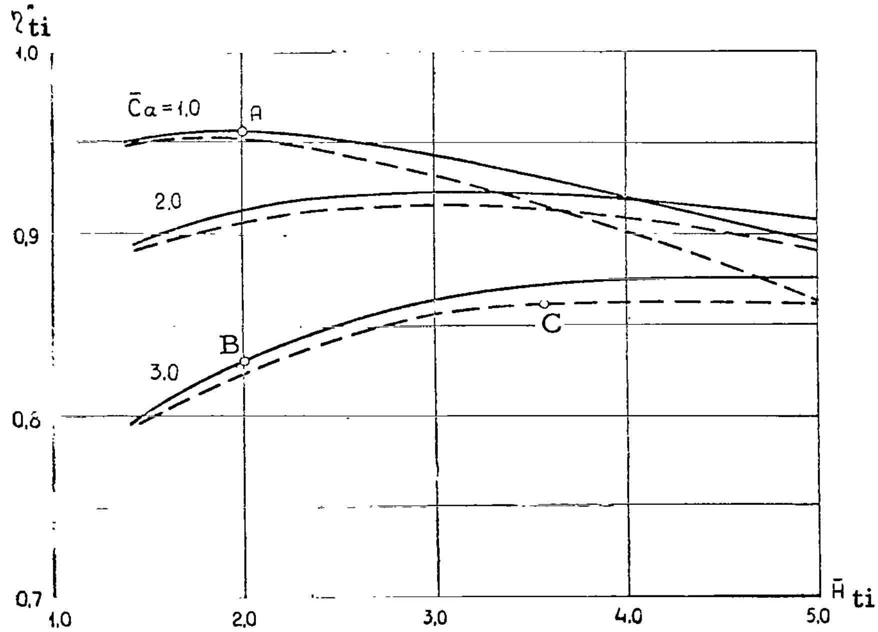

Figure 9. Evaluation of the Effectiveness of Rectifying Vanes Behind the Turbine:——Without Rectifying Vanes;– – – With Rectifying Vanes.

It is apparent that the requirement for obtaining a nearly axial discharge from the turbine means limiting the load factor to a value of $\overline{H}_{ti} \cong 2.0$ in the absence of rectifying vanes. For small values of the coefficient of discharge $\overline{C}_a = 1.0$, the design of such a turbine stage does in fact make sense (point A), since increasing $\overline{H}_{ti}$ lowers the efficiency of the turbine; it is lowered even further by the placement of rectifying vanes in order to obtain axial exhaust. If, for example, it is required for reasons of strength to have high values of $\overline{C}_a$ ($\overline{C}_a = 3.0$), the design of the last stage with $\overline{H}_{ti} = 2.0$ (to obtain axial exhaust) is unsuitable (point B). In this case, it is better to design the last stage with $\overline{H}_{ti} = 3.5$ and to use rectifying vanes (point B), since the efficiency is now higher.

5. CONCLUSION

The above analysis shows that in calculating turbines with working media characterized by a high velocity of sound (helium and hydrogen), it is advisable to choose higher values for the load factor and coefficient of discharge.

In this case, it is necessary in order to obtain axial exit flow from the last stage to place rectifying or directing vanes behind the turbine.

It must be pointed out, however, that in each specific instance these recommendations must be refined, depending on the resultant blade height and other structural, durability, and weight criteria of the given turbine.

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